Sensitivity Analysis of Vibration Signals for Controlled and Un-Controlled Factors for Symmetric and Asymmetric Number of Ball Supports in the Ball Bearing

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Abstract - This paper presents mathematical model of ball bearing to analyze the effect of controlled and un-controlled factors in case of rotating machines. Vibration signals are sensitive to controlling as well as un-controlled working factors. Number of balls present in the ball bearing also plays a significant role in the enhancement of the vibration amplitude. Number of balls in the bearing represents number of supports for the inner race. In this paper, experimental results of bearing with N_B=8 are compared with the results of mathematical model with N_B=8 and then changes have been made in the mathematical model to determine the effect of number of balls in the bearing for similar conditions. Sensitivity analysis is carried out for every model by using Taguchi's Methodology. The symmetric and asymmetric roller supports affects level of bearing vibrations. The bearing model with suitable number of balls can be selected as per need or application, depending upon the level of vibrations getting for certain working conditions.

Keywords: Condition Monitoring, rolling element bearing, localized defects.

I . INTRODUCTION

Rolling element bearings are used widely in all rotating machines. The accuracy of the machine output is very important. Early failure or abnormal behavior of the bearing affects accuracy of it. Every bearing has finite working life for a fixed load and running speed. But due to excessive loads premature fatigue occurs. Tight fits, brinelling and improper preloading can also bring about early fatigue failure. In such a case it is desire to reduce the load acting on it or redesign a bearing which will have greater capacity. This obviously will increase production time reducing the time required for remaining processes. Sometimes due to higher speeds and loads, temperature inside the bearing may rise above the limit and may affect the hardness causing early failure. In extreme cases ball and rigs may deform and sometimes lubricant inside the bearing may loss due to excess temperature. All these problems may arise due to improper or asymmetric loading; hence improper supports provided to the inner race which takes the load are indirectly responsible for premature fatigue failure, temperature rise due to friction and loss of lubricant in the bearing and misalignment. The symmetric supports helps to reduce the level of vibration generated due to speed and defects on the bearing elements. The level of vibration reduces when the number of supports increases [R K Purohit et al, 2006]. The variation in the number of balls may affect on the performance and life of that bearing.

II . MODELING OF ROLLING ELEMENT BEARING

The ball bearings play important role in rotating machineries. The behavior of the bearing can be understood very easily by using mathematical model. Mathematical model of the bearing helps to understand the stresses developed into the bearing, when loads are being applied by different elements of the system and the shaft or inner race of the bearing is rotating with certain speed.   In this the bearing model has been developed by considering the contacts between the balls and the races as the Hertzian contacts. The assumptions of mathematical model are as follows:

1. Inner race is rotating with the speed of shaft and Outer race is fixed with help of housing.
2. Revolving and rotating balls will act as a spring, and it gets compressed, whenever comes in the loaded region. Balls are equidistant and temperature of the surrounding is constant.
3. There is no slipping of the balls. The races are rigid and undergo only local deformation due to contact stresses. The deformation takes place according to Hertzian theory of elasticity.

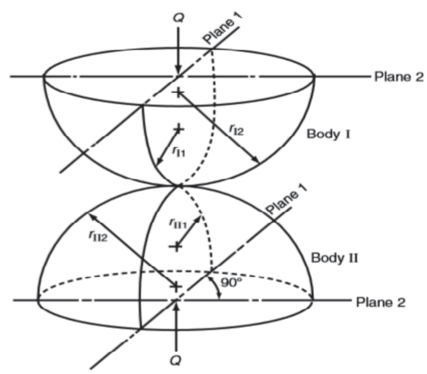


Figure 1 Model of the bearing [M S Patil et al, 2010]

The Hertzian forces acting at the contact point due to contact deformation, then it can be given as,

$$F = K\,\delta^{n} \qquad (1)$$

Where, F= force due to deformation, K= stiffness of the bearing, δ= contact deformation and n= 3/2 for ball bearing. To determine the force applied at the contact due to deformation, the stiffness of the ball bearing and contact deformation have been determined. The stiffness of the ball bearing mainly depends upon geometry of it.

r_{gi}= 4.2mm, r_{go} = 4.7mm, $f_o = (r_{go}/D) = 0.5875$, $f_i = (r_{gi}/D) = 0.525$

For contact in between ball and inner race and for contact in between ball and outer race,

$$r_{I1} = \frac{D}{2}, \quad r_{I2} = \frac{D}{2}, \quad r_{II1} = \frac{1}{2}d_o, \quad r_{II2} = r_{go}. \qquad (2)$$

Here,

D = Ball diameter, 8mm,

d_i= Inner Race Diameter, 25.2mm,

d_o = Outer Race Diameter, 41.8mm,

d_m= Pitch Diameter, 33.5mm,

r_{gi}= Inner Race Groove Diameter, 4.2mm,

r_{go}= Outer Race Groove Diameter, 4.7mm,

r= Radius of Curvature of body

r_{12}= radius of curvature of body 1 in plane 2.

$$\Sigma\rho = \text{curvature sum} = \{(1/r_{11})+(1/r_{12})+(1/r_{1I1})+(1/r_{1I2})\} = \rho_{I1} + \rho_{I2} + \rho_{II1} + \rho_{II2} \qquad (3)$$

$$F(\rho) = \text{Curvature difference} = \frac{(\rho_{I1}-\rho_{I2}) + (\rho_{II1}-\rho_{II2})}{\Sigma\rho} \qquad (4)$$

So from this one could get the values of δ^* i.e. the dimensionless contact deformation. Also the values of δ^* with the help of curvature difference $F(\rho)$ has been evaluated using table of $F(\rho)$ and δ^* [Hulugappa et al, 2012].

Total deflection between two raceways is the sum of the approaches between the rolling elements and each raceway. Here, K_i and K_o are inner and outer raceways to ball contact stiffness;

$$K = \left[\frac{1}{(1/K_i)^{1/n} + (1/K_o)^{1/n}}\right]^n \tag{5}$$

And this are given by

$$K_p = 2.15 \times 10^5 \sum \rho^{-1/2} (\delta^*)^{-3/2}, \tag{6}$$

Thus, the values of the curvature sum and contact deformation as;

For inner race, $\Sigma\rho$= 0.34033, $F(\rho)$=0.930, δ^*= 0.62415, so; K_{pi}=747403.0758 N/mm^2

For outer race, $\Sigma\rho$= 0.23901, $F(\rho)$=0.68913, δ^*= 0.86365, so; K_{po}=541928.3328N/mm^2

Hence, for SKF 6204 Ball Bearing, $K=7.097611554 \times 10^9$ N/m$^{3/2}$

Considering the damping constant for steel material ball bearing as, C=200Ns/m. and considering the initial conditions that are obtained with the help of geometry of the ball bearing, the characteristic defect frequencies have been determined [M S Patil et al, 2010].

Fundamental Train Frequency, $F_{FTF} = \frac{N_s}{(2\times 60)}\left[1 - \frac{D}{d_m}\cos\alpha\right]$ (7)

Inner Race Frequency, $F_{ID} = N_B \times \left[\frac{N_s}{60} - F_{FTF}\right]$ (8)

Outer Race Frequency, $F_{OD} = Z \times F_{FTF}$, (9)

Ball Spin Frequency, $F_B = \frac{N_s}{(2\times 60)}\frac{d_m}{D}\left[1 - \left(\frac{D}{d_m}\right)^2 \cos\alpha\right]$ (10)

Cage Speed, $\omega_C = \left(\frac{2\pi N_s}{60\times 2}\right)\left[1 - \frac{D}{d_m}\cos\alpha\right]$ (11)

If x and y are the deflections along X- and Y-axis and Cr is the internal radial clearance, the radial deflection at the i^{th} ball, at any angle yi is given by,

$$\delta = [(x\cos\theta_i + y\sin\theta_i) - C_r] \tag{12}$$

The mathematical model is prepared by considering that, the balls in the loaded region will give the restoring force. These springs will give the restoring force only when are compressed. The deflection of the bearings is considered as X and Y along X axis and along Y axis. So in order to evaluate restoring force, the clearance in the bearing should be neglected. Therefore the restoring force after contact deformation along X and Y direction can be given as,

$$F_{XX} = [(x\cos\theta_i + y\sin\theta_i) - C_r]^{\frac{3}{2}}\cos\theta_i \tag{13}$$

$$F_{YY} = [(x\cos\theta_i + y\sin\theta_i) - C_r]^{\frac{3}{2}}\sin\theta_i \tag{14}$$

This restoring force will be the force acting by the balls which are in the loading region and for non-defective bearing. But when there is defect in the bearing having height, H_D, and φ, the angle made by the defect with the centre of the bearing, which is given by ratio of defect size to raceway radius, the total deflection occurred into the bearing will consider the clearance as well as the height of the defect, i.e.

$$F_{XX} = \sum_{i=1}^{Z}\left[(x\cos\theta_i + y\sin\theta_i) - \left(C_r + H_D\sin(\pi(\theta_i - \theta_i)/\varphi)\right)\right]^{\frac{3}{2}}\cos\theta_i \quad (15)$$

$$F_{YY} = \sum_{i=1}^{Z}\left[(x\cos\theta_i + y\sin\theta_i) - \left(C_r + H_D\sin(\pi(\theta_i - \theta_i)/\varphi)\right)\right]^{\frac{3}{2}}\sin\theta_i \quad (16)$$

Since outer race is stationary, $\theta_i = \omega_c t + 2\pi(Z - i)/Z$ and (17)

as inner race is rotating with the shaft, $\theta_i = (\omega_c - \omega)t + 2\pi(Z - i)/Z$ (18)

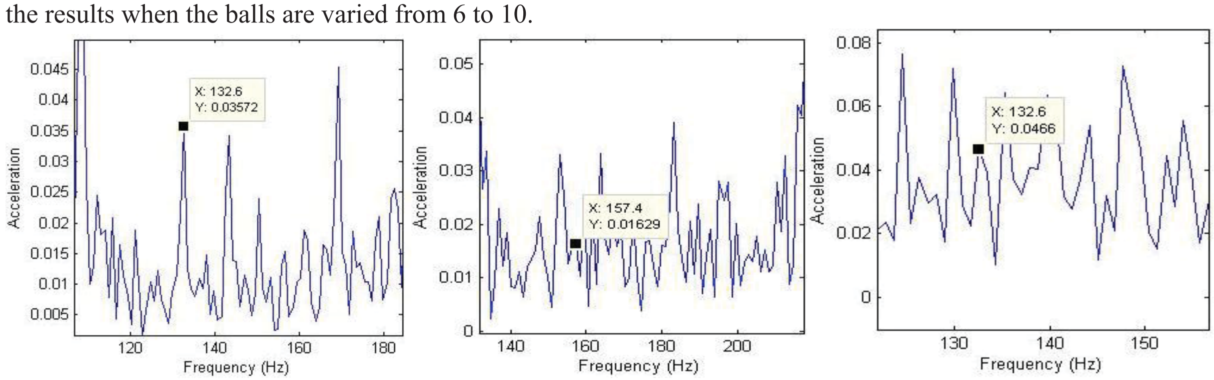
The equation of motion for two degree of freedom in X and Y direction is given by,

In X direction, $M\ddot{x} + C\dot{x} + F_{XX} = W$ and in Y direction $M\ddot{y} + C\dot{y} + F_{YY} = 0$ (19)

The depth of the defect for finding out the displacement of the centre of the ball when it goes into the spall or defect is evaluated using the relation, Depth= R - (R cosØ) where Ø is the angle inscribed by the defect width at the centre of the ball rotating in the bearing and R is the radius of the ball. In order to find out the sensitiveness of factors and their levels, Taguchi's methodology has been considered. The information which factor and level are significant is possible to determine using it. L8 orthogonal array is used for analysis of the vibration signals. The vibration amplitude presented here is the amplitude obtained using FFT of the Acceleration signals. The FFT is obtained from acceleration signals thus all the features of the vibration signals evaluated from the bearing model have been considered here.

III . RESULTS AND DISCUSSION

The vibration signals extracted from the bearing model are represented below. The results represented here are for model having 8 balls in the bearing. The results have been compared with the results that are obtained by experimentation using LABVIEW 2012 and NI Instruments (Accelerometer). The changes have been made in the bearing model to consider the effect of the varying number of balls from 6 to 10. The level of vibration signals obtained in such manner varies as conditions for the experiments are varying. The variation has been observed in the results when the balls are varied from 6 to 10.



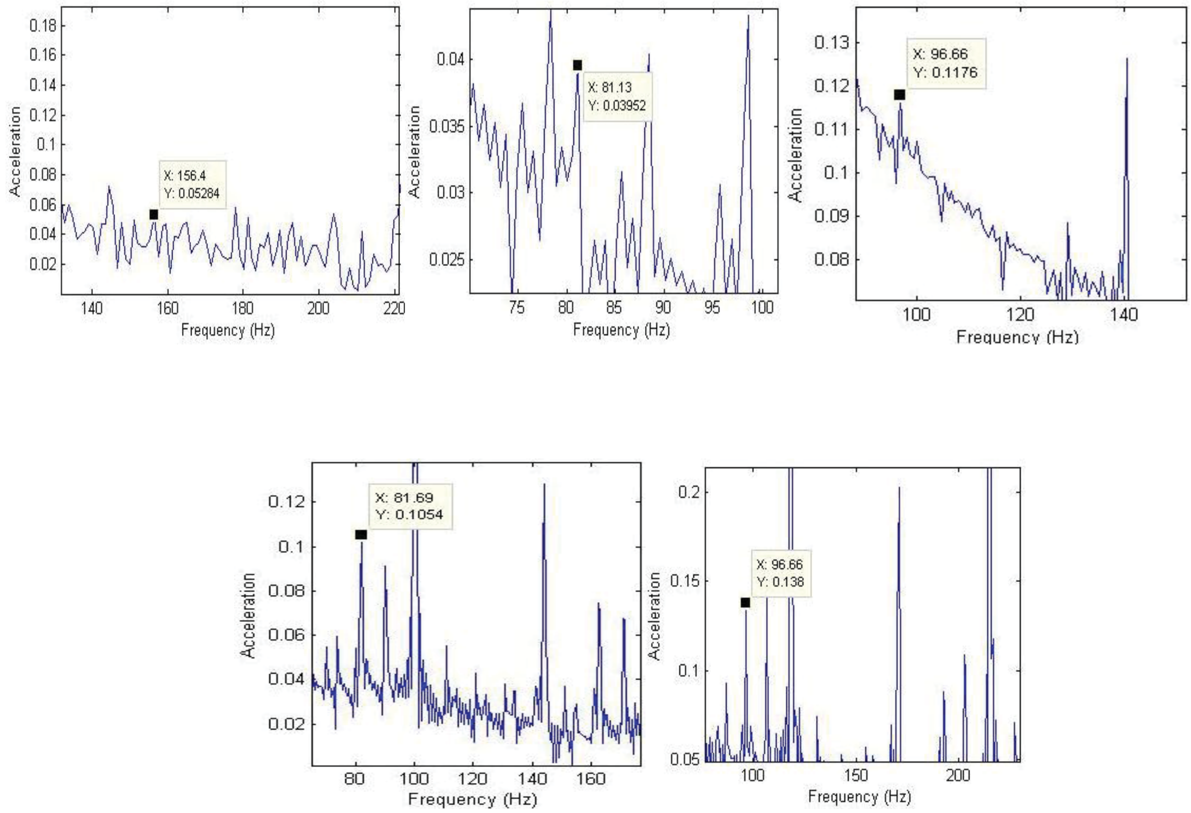


Figure 2. Fast Fourier Transform of the vibration (acceleration) signals calculated from the model.

The results represented are for Nb=8. The results for Nb=6,7,9 and 10 are considered only for analysis of the sensitivity of the vibration signals. Sensitivity analysis of the results obtained for different number of balls shows mixed results. The level of vibration signals for a model having Nb=6 is lesser than other models, and it may be due to the fact that in Nb=6.s

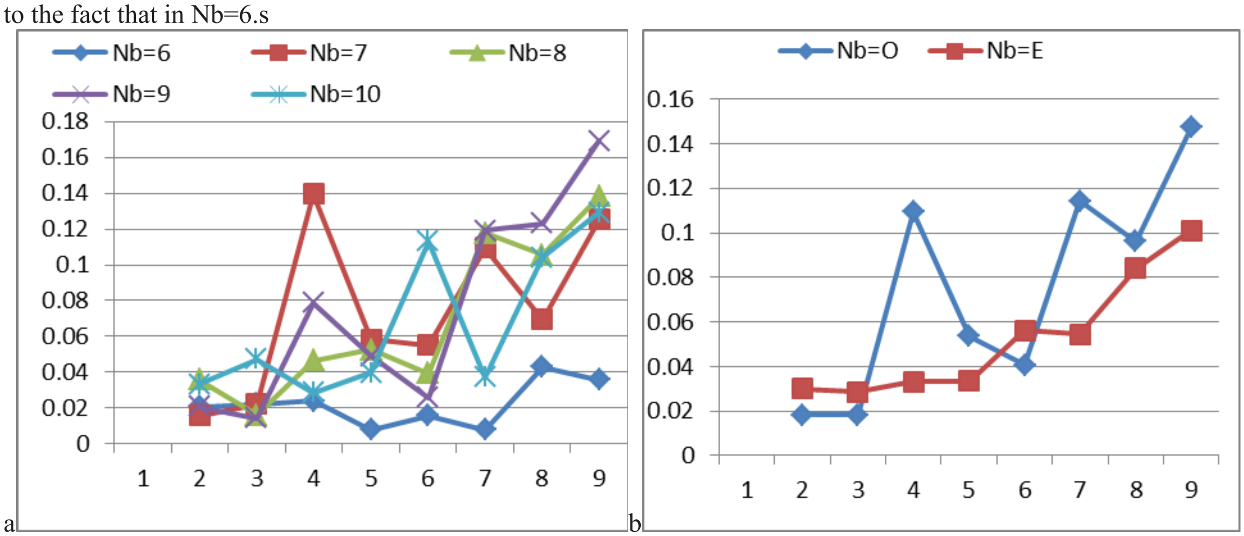
a  b

Figure 3 a. Acceleration results for Nb=6 to Nb=10 b. Acceleration results for Odd and Even Balls

Here it can be seen from the figure 2.a, the vibration level for the models with Nb=6 are comparatively lesser than other models. But model with Nb=7 shows almost opposite behavior than that of Nb=6. It is very difficult to predict or decide about suitable number of balls for a rolling element bearing from the obtained results. It is well known fact that, symmetric supports gives more stability and reduces vibrations of any system as these supports shares load equally and balances the system. Thus even number of balls in the bearing must helps to reduce the vibrations that are generated in defective or non defective bearing system. In the rolling element bearing, even number of balls will be available in the lower region, when total number of balls in the bearing will be even. Thus the average values for odd and even number of balls are considered for finding out suitable number of balls for the bearing system. For this, average values are considered for odd number of balls (Nb=7,9) and even balls (Nb=6,8,10) to check the effect of symmetric and asymmetric supports in the bearing.It is very clear that, as defect size progresses, vibration inside the bearing increases. It also have been seen that vibrations due to defect increases, when the defect comes on outer race. Figure 2.b shows that, even number of supports helps to reduce vibrations inside the bearing. When odd number of balls are present in the bearing, the system will become unstable and hence the vibrations due to defect will enhance the level of vibrations. It mainly shows that, symmetric type of supports helps in reducing vibrations generated due to defect inside rolling element bearing.

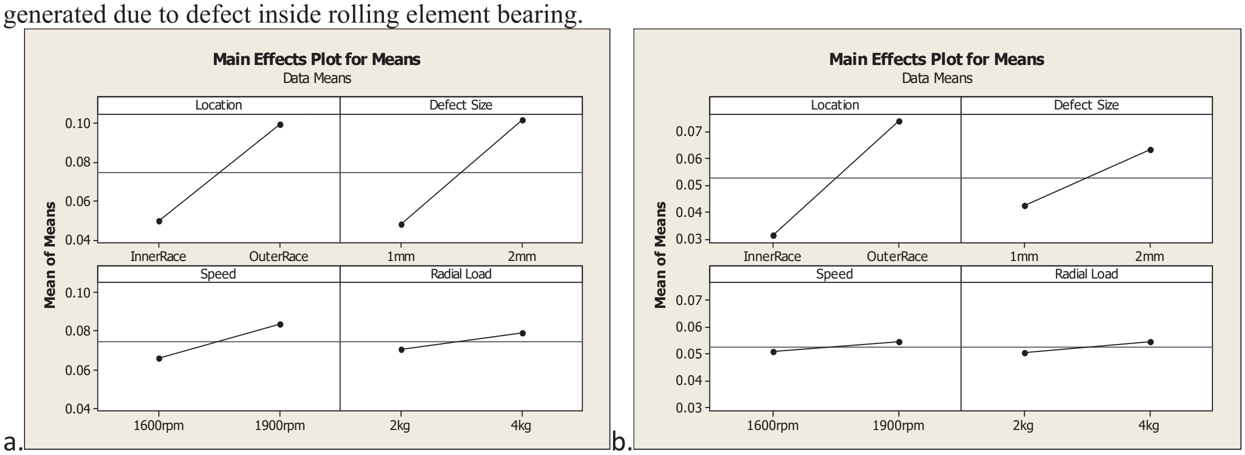


Figure 4. Response of the bearing system to Odd (a) and Even (b) number of Ball supports

Above Figure 4.a and 4.b shows the response of Taguchi analysis for odd and even number of balls. When numbers of balls in the bearing are odd, the defect size comes out to be most significant factor that affects vibration signal level and defect location is the second most significant factor. Outer race defect enhances level of vibration signals as compared to inner race defects.

IV . CONCLUSION

The vibration signals of a rolling element bearing are sensitive to the defect location, defect size, rotating speed of the shaft or inner race and the radial load acting on the bearing system. Amplitude of vibration signals varies as the working factors are varied. It is found that when the ball touches the defect a peak is obtained in the frequency spectrum of the system. When the defect was present on the inner race, a peak was observed in the response of the system at the characteristic frequency of the inner race. Similar results are obtained when the defect is on outer race.

It is found that, from the comparison between frequency spectrums of the bearings having 6, 7, 8, 9 and 10 number of balls, the bearing with Nb = 6 shows lowest level of amplitude of vibration signals. It means that Nb = 6 is the optimum number of ball supports for minimum vibrations in the rolling element bearing. It may be because of the angle between adjacent balls and the effective load acting on the lower region balls in the bearing. As the angle is greater (60o), the cosine component of the vertically downward radial load acts on the lower region balls and hence, the peak obtained in bearing due to Nb= 6 is lesser than Nb = 10. In case of a bearing with odd (asymmetric) number of ball supports, defect size is the most dominant factor which hampers the vibration signals of the system and in case of a bearing having even (symmetric) number of ball supports, it is found that the defect location is the most significant factor affecting the system vibrations. Since these both parameters are not controllable so by manipulating them system vibrations can be reduced. Remaining parameters i.e. rotating speed of the shaft and the radial load acting on the system do not contribute too much to hamper the system vibrations in both the cases. So while designing the rolling element bearing, symmetric (even) ball supports must be considered to avoid the breakdown of the bearing system by reducing its vibrations.

Maintenance of the system can be done by predicting about the bearing as faulty or not, based on the results obtained. Future research can also focus on prediction of the remaining useful life of the bearing after defect is detected in the bearing. One can go for the analysis of large domain to determine the most significant factor and effect of various working factors on the vibrations of the system. Also ball defects can be considered for finding the effects of parameters on the vibration signals.

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